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27 NOV 2003

Mail Stop Non-fee Amendment
Commissioner for Patents
P.O. Box 1450
Alexandria, VA 22313-1450

Re: Inventor: Robert Louis Giuliani
Application no. 10/643274
Filing date: 08/18/2003
Title: Interchangeable 2-stroke or 4-stroke High Torque Power Engine
CIP of application no.10/252,927 filing date 09/24/2002
Art Unit: 3748
Confirmation no. 4067

INTRODUCTORY COMMENTS

The attached "Amendments to the Specification" of the above application no.10/643274 are on the following sheets. The section to be amended is on sheets 2,3. The amendments for it are on sheets 4-7. The REMARKS are on sheet 8.

I've tried to make this amendment comply with "Waiver of 37 CFR 1.121" "Revised Amendment Format" that I took from the internet. If it does not agree with "Waiver of 37 CFR 1.121" "Revised Amendment Format", please let me know where my mistakes are and I will correct them. I can be reached at my above address, email or phone no. If by phone, the best time to reach me is 7:30AM – 8:30AM, Hawaii time. I think that the East Coast is 5 hours ahead of Hawaii time.

There are no Amendments to the Claims.

R.L. Giuliani

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Email: rlgiuliani@earthlink.net

~~Underlying Mathematics.~~

~~The discussion below references the following equations by their definitions, e.g. F lbf. Some Complete equations are also included in the discussion.~~

~~Definitions:~~

~~1 BTU = 778 ft-lbf~~

~~1 hp = 550 ft-lbf/sec~~

~~$2\pi r'$ = length of 1-way clutch rim at connecting rod contact. (ft).~~

~~F – actual mean combustion force/piston (lbf)~~

~~F′ – most efficient mean combustion pressure/piston (lbf/in^2)~~

~~Fr – fuel flow rate (lbm/sec.)~~

~~hp – shaft horsepower (1 hp = 550 ft-lbf/sec)~~

~~Lo – Power losses (fraction of hp)~~

~~n – total number of pistons. 2,4,6, ...~~

~~n/2 – 2 stroke. Number of equally spaced overlapping pistons cycling through the power stroke.~~

~~$n^2/2$ – 2-stroke shaft power. (ft-lbf/sec).~~

~~n/4 – 4-stroke. Number of equally spaced overlapping pistons cycling through the power stroke.~~

~~$n^2/4$ – 4-stroke shaft power. (ft-lbf/sec) See FIG 6.~~

~~Qe – fuel's energy density. (BTU/lbm).~~

~~r′ – 1-way clutch radius at connecting rod contact. (ft). See FIG 15.~~

~~ra – radius of cylinder. (in)~~

~~Rv – power shaft's rotation rate. (rpm)~~

~~Sp – shaft power + losses. (ft-lbf/sec.)~~

~~T – Torque. (lbf-ft)~~

~~Vp – piston's velocity. (ft/sec)~~

~~Equations:~~

~~$Vp = (2\pi)(r')(Rv)/(60)$ Piston's speed and the 1-way clutch rim speed are equal at contact..~~

~~$r' = 60(Vp)/2\pi(Rv) = 30(Vp)/\pi(Rv)$ r′ is central to this engine's design and operation..~~

~~$Rv = 30(Vp)/\pi r'$~~

~~$Sp = 550(hp)(1 + Lo)$~~

~~$Fr = (Sp)/(778Qe)$~~

~~$Fr = (F)(n^2)(Vp)/[2(778)(Qe)]$~~

~~$F = 2Sp/(n^2Vp)$~~

~~$T = F(r')$~~

$\cancel{F' = F/(\pi r_b^2)}$

$\cancel{r_b^2 = F/(\pi F')}$

$\cancel{\text{bore} = 2\sqrt{F/(\pi F')}}$

~~The advantage of overlap is evident in the next two examples that compare the number of cylinders in this smaller engine with the number of cylinders in a crank engine of equal power. The examples also show the power advantage of this engine's overlapping 2-stroke over its 4-stroke.~~

~~1. Example of this 2-stroke engine with **n** cyls. vs the number of crank engine cyls. of equal power:~~

~~Let **n** = 6 then~~ $\cancel{\mathbf{n}^2/2 = 18}$ ~~crank engine cyls.~~

~~2. Example of this 4-stroke engine with **n** cyls. vs the number of crank engine cyls of equal power:~~

~~Let **n** = 8 (two banks of 4 pistons each in FIG 8) then~~ $\cancel{\mathbf{n}^2/4 = 16}$ ~~crank engine cyls.~~

~~The deactivation feature also makes a 4-Strokebank combined with 2-Stroke pairs advantageous.~~

First, consider the benefit of overlapping power pistons on the power stroke e.g., a 2-stroke, 6 cyl engine with a 9" piston stroke would simultaneously have the 1st piston 6" after tdc, the 2nd piston 3" after tdc and the 3rd piston igniting at tdc. The 6 pistons continuously cycle through their power strokes in this sequence. The power added by the 3rd piston is reduced by the combined remaining power of the 1st and 2nd pistons resulting in fuel savings and smooth power shaft rotation.

Underlying Mathematics.

Definitions:

1 BTU = 778 ft-lbf

1 hp = 550 ft-lbf/sec.

2πr' = length of 1-way clutch rim at connecting rod contact. (ft)

bore – cylinder diameter. (in.)

Cp – cylinder pressure calculated from known bore size. (psi)

F – combustion force per piston. (lbf)

F' – estimated combustion pressure per piston. Used to find the bore size. (psi)

Fr – fuel flow rate (lbm/sec)

hp – shaft horsepower.

k = 2 or 4 (**k** = 2 for a 2-stroke. **k** = 4 for a 4-stroke.)

Lo – power losses (fraction of hp)

n – number of active pistons. 2,4,6,8, ...

n/k – number of overlapping pistons cycling through the power stroke.

Qc – fuel's energy density. (BTU/lbm)

r' – 1-way clutch radius at connecting rod contact. (ft)

r – radius of cylinder. (in)

Rv – power shaft's rotation rate. (rpm)

Sp – shaft power + losses. (ft-lbf/sec)

T – torque per piston. (lbf-ft)

T' – total shaft torque. (lbf-ft)

Vp – piston velocity. (ft/sec)

Equations:

$\mathbf{Vp = \pi(r')(Rv)/(30)}$ Piston rod's speed and the 1-way clutch rim speed are equal at contact.

$\mathbf{r' = 30(Vp)/\pi(Rv)}$ **r', Vp, Rv** are central to this engine's design and operation.

$Rv = 30(Vp)/(\pi r')$

$F = 550hp(k)/(nVp)$

$F = 16500(hp)(k)/\pi(n)(Rv)(r')$

$T = F(r')$

$T' = nT/k$

$F' = F/[\pi(r^2)]$

$r^2 = F/(\pi F')$

$bore = 2[F/(\pi F')]^{.5}$

$Cp = 4F/(\pi bore^2)$

$F = \pi F'(bore^2)/4$

$Sp = 550hp(1+Lo)$

$Fr = (Sp)/(778Qc)$

$Fr = (F)(n)(Vp)/[k(778Qc)]$

Examples that find preliminary information to any size engine with a hand calculator. (800 psi is estimated where used.)

Example: 6 cylinder, 2-stroke, 700 hp.

1. Let: **hp** = 700; **Vp** = 15 ft/sec; **F′** = 800 psi; **n** = 6; **k** = 2; **r′** = .75 ft = 9 in.

   F = 2(700)(550)/[(6)(15)] = 8556 lbf

   Rv = 30(15)/(.75π) = 191 rpm

   bore = $2[(8556/800\pi)]^{.5}$ = 3.690 in.

   T = 8556(.75) = 6417 lbf-ft

   T′ = 6(6417)/(2) = 19251 lbf-ft

Example: 6 cylinder, 2-stroke, 1200 hp.

2. Let: **hp** = 1200; **Vp** = 22 ft/sec; **n** = 6; **k** = 2; **r′** = .75 ft. = 9 in. (Compare results to 1.)

   F = 2(1200)(550)/[(6)(22)] = 10000 lbf

   Rv = 30(22)/(.75π) = 280 rpm

   bore = 3.690 in. (from example 1.)

   Cp = $4(10000)/[\pi(3.690^2)]$ = 935 psi (Compare to **F′** = 800 psi in 1.)

   T = 10000(.75) = 7500 lbf-ft

$T' = 6(7500)/2 = 22500$ lbf-ft

Example: 8 cylinder, 4-stroke (2 banks of 4 cyls. each) 1200 hp engine. See FIG 6.

3. Let: **hp** = 1200; **F′** = 800 psi; **n** = 8; **k** = 4; **Rv** = 115 rpm; **r′** = 1.25 ft. (1 cyl. per 1-way clutch requiring eight 1-way clutches. 50% overlap.)

$Vp = 1.25\pi(115)/30 = 15.05$ ft/sec.

$F = 4(550)(1200)/[(8)(15.05)] = 21927$ lbf.

bore $= 2[(21927/800\pi)]^{.5} = 5.907$ in. (Compare to example 3.)

$T = 21927(1.25) = 27409$ lbf-ft

$T' = 8(27409)/4 = 54818$ lbf-ft.

Example: 4 cylinder, 4-stroke 200 hp automobile engine. See FIG 6

4. Let: **hp** = 200; **F′** = 800 psi; **n** = 4; **k** = 4; **Vp** = 15 ft/sec; **r′** = .5 ft = 6 in. (2 cyls. per 1-way clutch requiring two 1-way clutches. No power stroke overlap).

$F = 200(550)(4)/[(4)(15)] = 7333$ lbf.

$Rv = 30(15)/(.5\pi)$ 286 rpm.

bore $= 2[(7333/800\pi)]^{.5} = 3.416$ in.

$T = 7333(.5) = 3667$ lbf-ft.

$T' = 4(7333)/4 = 7333$ lbf-ft

Example: 8 cylinder, 2-stroke, 8,000 hp large marine engine.

5. Let: **hp** = 8000; **F′** = 800 psi; **n** = 8; **k** = 2; **Vp** = 28 ft/sec; **Rv** = 120 rpm. (1 cyl. per 1-way clutch requiring eight 1-way clutches. 14" piston stroke. 75% power stroke overlap.)

$F = 2(550)(8000)/[(8)(28)] = 39286$ lbf

$r' = 30(28)/(120\pi) = 2.228$ ft. The transmitting units 89 (FIGs 7,8) could be carried by a short outer race 5 with a single spoke 35 to reduce inertia.

bore $= 2[(39286/800\pi)]^{.5} = 7.907$ in.

$T = 39286(2.228) = 87529$ lbf-ft

$T' = 8(39286)/2 = 157144$ lbf-ft.

Next, comparing the number of cylinders in this smaller engine to the number of cylinders in an equal powered crank engine.

1. For a 2-stroke engine with **n** cyls., let **n** = 6 then $\mathbf{n}^2/2$ = 18 crank engine cyls.

2. For a 4-stroke engine with **n** cyls., let **n** = 8 (two banks of 4 pistons each in FIG 6), then $\mathbf{n}^2/4$ = 16 crank engine cyls.

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Confirmation no. 4067

REMARKS

1. In the above application no.10/643274, delete all the words in the entire section between the heading "DETAILED DESCRIPTION OF THE INVENTION" and the subheading "**Discussion.**". A copy of these words is shown on pgs 2 and 3 of this amendment with a line drawn through them.

2. Replace all the deleted words by inserting all the underlined words on pgs 4-7 of this amendment entirely between the heading "DETAILED DESCRIPTION OF THE INVENTION" and the subheading "**Discussion.**". That will complete this amendment.

3. This amendment is primarily caused to clarify the mathematics. Only a few of the equations need to be changed but changing them individually greatly increases the chances for errors so I decided to use the method in this amendment. The examples are only to show how the math can be used to find this engine's primary values for any size engine with a hand calculator.

4. I included the first paragraph in the amendment to expand upon the original page 8, lines 7-9.

5. There is no new matter in this amendment.

6. If there are questions, please contact me at my ADDRESS, RLG email or phone number. If by phone, the best time to call is 0730-0830 Hawaii time.

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INTRODUCTORY COMMENTS

This is the 2nd amendment to this CIP application no. 10/643274. The 1st amendment was dated 27 NOV 2003. Replace the entire 27 NOV 2003 amendment and more with this 2nd amendment.

See the REMARKS on sheet 15.

I've tried to make this 2nd amendment agree with "Revised Amendment Practice Effective July 30, 2003" that I took from the internet. The following sheets for "Amendments to the specification:" are my best effort to avoid errors by replacing all the lines between the CIP section heading DETAILED DESCRIPTION OF THE INVENTION and the subsection heading **Interchanging 4-Stroke and 2-Stroke**. in the CIP. If it is unclear, please let me know where my mistakes are and I will correct them. I can be reached by email or phone no. If by phone, the best time to reach me is 7:30AM – 8:30AM, Hawaii time.

Unlike all my earlier patent applications, this has been a very difficult engine invention to put into an application because it is a great departure from existing engines, which opens many ramifications that I'm trying to describe and claim. That is what causes the CIP and two amendments to the CIP.

R.L. GIULIANI
Inventor/Applicant

~~First, consider the benefit of overlapping power pistons on the power stroke e.g., a 2-stroke, 6 cyl engine with a 9" piston stroke would simultaneously have the 1st piston 6" after tdc, the 2nd piston 3" after tdc and the 3rd piston igniting at tdc. The 6 pistons continuously cycle through their power strokes in this sequence. The power added by the 3rd piston is reduced by the combined remaining power of the 1st and 2nd pistons resulting in fuel savings and smooth power shaft rotation.~~

~~Underlying Mathematics.~~

~~Definitions:~~

~~1 BTU = 778 ft-lbf~~

~~1 hp = 550 ft-lbf/sec~~

~~2πr′ = length of 1-way clutch rim at connecting rod contact. (ft).~~

~~**bore** – cylinder diameter. (in.)~~

~~**Cp** – cylinder pressure calculated from know bore size. (psi)~~

~~**F** – combustion force perpiston (lbf)~~

~~**F′** – estimated combustion pressure per piston. Used to find bore size. (psi)~~

~~**Fr** – fuel flow rate (lbm/sec.)~~

~~**hp** – shaft horsepower~~

~~k = 2 or 4 (k = 2 for a 2-stroke. k = 4 for a 4-stroke.)~~

~~**Lo** – Power losses (fraction of hp)~~

~~**n** – number of active pistons. 2,4,6, 8, ...~~

~~**n/k** – number of overlapping pistons cycling through the power stroke.~~

~~**Qc** – fuel's energy density. (BTU/lbm).~~

~~**r′** – 1-way clutch radius at connecting rod contact. (ft)~~

~~**r** – radius of cylinder. (in)~~

~~**Rv** – power shaft's rotation rate. (rpm)~~

~~**Sp** – shaft power + losses. (ft-lbf/sec.)~~

~~**T** – Torque per piston. (lbf-ft)~~

~~**T′** – total shaft torque. (lbf-ft)~~

~~**Vp** – piston's velocity. (ft/sec)~~

~~Equations:~~

~~**Vp = (π)(r′)(Rv)/(30)** Piston's speed and the 1-way clutch rim speed are equal at contact..~~

~~**r′ = 30(Vp)/π(Rv)** **r′,Vp,Rv** are central to this engine's design and operation..~~

~~**Rv = 30(Vp)/πr′**~~

~~**F = 550hp(k)/(nVp)**~~

~~$F = 16500(hp)(k)/\pi(n)(Rv)(r')$~~

~~$T = F(r')$~~

~~$T' = nT/k$~~

~~$F' = F/[(\pi r_b^2)]$~~

~~$r^2 = F/(\pi F')$~~

~~$bore = 2[F/(\pi F')]^{.5}$~~

~~$Cp = 4F/(\pi bore^2)$~~

~~$F = \pi F'(bore^2)$~~

~~$Sp = 550hp(1+Lo)$~~

~~$Fr = (Sp)/(778Qe)$~~

~~$Fr = (F)(n)(Vp)/[k(778Qe)]$~~

~~Examples that find preliminary information to any size engine with a hand calculator. (800 psi is estimated where used.)~~

~~Example: 6 cylinder, 2 stroke 700 hp.~~

~~1. Let: **hp** = 700; **Vp** = 15 ft/sec; **F′** = 800 psi; **n** = 6; **k** = 2; **r′** = .75 ft = 9 in.~~

~~**F** = 2(700)(550)/6(15) = 8556 lbf.~~

~~**Rv** = 30(15)/(.75π) = **191** rpm~~

~~**bore** = 2(8556/800π)$^{.5}$ = 3.690 in.~~

~~**T**. = 8556(.75) = 6417 lbf-ft~~

~~**T′** = 6(6417)/(2) = 19251 lbf-ft.~~

~~Example: 6 cylinder, 2 stroke, 1200 hp.~~

~~2. Let: **hp** = 1200; **Vp** = 22 ft/sec; **n** = 6; **k** = 2; **r′** = .75 ft. = 9 in. (Compare results to 1.)~~

~~**F** = 2(1200)(550)/[(6)(2)] = 10000 lbf.~~

~~**Rv** = 30(22)/(.75π) = 280 rpm~~

~~**bore** = 3.690 in. (from example 1.)~~

~~**Cp** = 4(10000)/[π(3.690^2)] = 935 psi. (Compare to **F′** = 800 psi in 1.)~~

~~**T** = 10000(.75) = 7500 lbf-ft~~

~~**T′** = 6(7500)/2. = 22500 lbf-ft~~

~~Example: 8 cylinder, 4 stroke (2 banks of 4 cyls. each) 1200 hp engine. See FIG 6.~~

~~3. Let: **hp** = 1200; **F′** = 800 psi; **n** = 8; **k** = 4; **Rv** = 115 rpm; **r′** = 1.25 ft. (1 cyl. per 1-way clutch requiring eight 1-way clutches. 50% overlap.)~~

~~**Vp** = 1.25π(115)/30 = 15.05 ft/sec.~~
~~**F** = 4(550)(1200)/[(8)(15.05)] = 21927 lbf.~~
~~**bore** = 2(21927/800π)$^{.5}$ = 5.907 in. (Compare to example 3.)~~

~~$T = 21927(1.25) = 27409$ lbf ft.~~

~~$T' = 8(27409)/4 = 54818$ lbf ft.~~

~~Example: 4 cylinder, 4-stroke 200 hp automobile engine. See FIG 6.~~

~~4. Let: **hp** = 200; **F′** = 800 psi; **n** = 4; **k** = 4; **Vp** = 15 ft/sec; **r′** = .5 ft. = 6 in. (2 cyl. per 1-way clutch requiring eight 2-way clutches. No power stroke overlap.)~~

~~**F** = $4(550)(200)/[(4)(15)] = 7333$ lbf.~~

~~**Rv** = $30(15)/(.5\pi) = 286$ rpm.~~

~~**bore** = $2(7333/800\pi)^{.5} = 3.416$ in.~~

~~**T** = $7333(.5) = 3667$ lbf ft.~~

~~**T′** = $4(7333)/4 = 7333$ lbf ft.~~

~~Example: 8 cylinder, 2-stroke, 8,000 hp large marine engine.~~

~~5. Let: **hp** = 8000; **F′** = 800 psi; **n** = 8; **k** = 2; **Vp** = 28 ft/sec; **Rv** = 120 rpm. (1 cyl. per 1-way clutch requiring eight 1-way clutches. 14" piston stroke. 75% power stroke overlap.)~~

~~**F** = $2(550)(8000)/[(8)(28)] = 39286$ lbf.~~

~~**r′** = $30(28)/(120\pi) = 2.228$ ft. The transmitting units 89 (FIGs 7,8) could be carried by a short outer race 5 with a single spoke 35 to reduce inertia.~~

~~**bore** = $2[(39286/800\pi)]^{.5} = 7.907$ in.~~

~~**T** = $39286(2.228) = 87529$ lbf ft.~~

~~**T′** = $8(39286)/2 = 157144$ lbf ft.~~

~~Next, comparing the number of cylinders in this smaller engine to the number of cylinders in an equal powered crank engine.~~

~~1. For a 2-stroke engine with **n** cyls., let **n** = 6 then $\mathbf{n}^2/2 = 18$ crank engine cyls.~~

~~2. For a 4-stroke engine with **n** cyls., let **n** = 8 (two banks of 4 pistons each in FIG 6),~~

~~then $\mathbf{n}^2/4 = 16$ crank engine cyls.~~

~~**Discussion**.~~

~~A pair of combustion cylinders 33 and related pairs of parts that include a pair of 1-way clutches (FIGs 1-3) make the basic 2-stroke engine in this invention. The clutch's inner race 4 is keyed to the power shaft 8. The outer race 5 carries a sector gear 12. Each gear 12 engages an opposite side of idler 40 whereby synchronous reverse motion is transmitted between the power piston 38 and the second piston 38 in the pair as the inner race 4 transmits the power to the shaft 8. Moving parts that are not shown with arrows 42 are presumed obvious.~~

~~Combining two pairs with idler 40A creates a 4-stroke shown in FIG 6 that will be described later under Interchanging 4-stroke and 2-stroke.~~

~~FIG 2 shows a gear mesh to transmit the piston's power between piston rod 18 and the outer race 5 of the 1-way clutch. Rod 18 reciprocates along a straight path 42. FIG 2 also shows a reciprocating starter 46 gear meshed with the outer race 5. By shifting race 5, the starter shifts both pistons 38 until ignition. Alternatively, shaft 43 can be used to shift the pistons until ignition. The 4-stroke version in FIG 6 needs one starter 46 (not shown).~~

~~One end of a V-belt or a chain 9 is fastened to the outer race 5 (FIGs 1,3). The way it is wrapped around race 5 always keeps it taut, which prevents backlash as it rotates race 5 in response to the power stroke. Rod 18 is connected to the other end of the belt or chain 9 with a suitable fastener 41.~~

~~The 1-way clutch's override feature in this engine allows output shaft 8 and the clutch's inner race 4 to rotate independently of the pistons 38 when the inner race's speed is greater than the outer race 5 speed. This feature creates regenerated energy is collection in an energy storage device 26 (FIG 5) available, e.g. for dumping to shaft 8 on demand or generating electricity.~~

~~The fixed length torque arm 10 (FIGs 2,3) causes instant peak torque at the beginning of the power stroke. A connecting rod guide 21, secured to housing 15, eliminates side thrust and reduces wear by keeping the piston 38 square in its cylinder. Wrist pins and piston skirts are not needed. The guide 21 is combined with a decelerator mechanism (FIG 4) to stop piston 38 at or near top dead center. The decelerator includes a node 19 that is part of each rod 18 in a pair and a spring 45 for each node. The spring is encased in the guide 21. An opening in the housing 15 allows easy replacement of the spring. The spring absorbs the impact of node 19 to halt the motion of piston 38, which is then accelerated on its power stroke by timely expanding combustion gases. The impact is reduced because node 19 is decelerating due to the power loss of the power piston to the shaft 8. The decelerator is positioned to prevent backlash of the gears 12 (FIGs 1,6) that mesh with idler 40.~~

~~A computer 7 (FIG 5) monitors input from the throttle 6 and shaft power from the sensor 22 on shaft 8 through leads 23 to determine the size of the combustion charge (**Fr** = (Sp)/(778)Qc)lbm/sec) to transmit to the cylinders through injector lines 24. The position of piston 38 is monitored through sensors 22 on shaft 43 and used for ignition timing. By monitoring the motion of each shaft 43 in several pairs, the computer controls timing between the unconnected pairs in a 2-stroke embodiment. The computer begins a power stroke with a piston in one pair when a piston in another pair is partly through its power stroke. In a 2-stroke, 50% power stroke overlap and smooth rotation of the shaft 8 is had with two unconnected pairs (four cylinders). Greater overlap is gained with more pairs.~~

~~Small Flywheels.~~

~~Load changes on shaft 8 could decrease **F** lbf below what is needed for an efficient combustion pressure. A small, suitable flywheel 48 is splined to the end of shaft 43 (FIG 3) to briefly increase chamber pressure for a more complete combustion with decreased emissions. Then, it dispenses the regenerated energy that it gains to moderate the speed of the pistons 38. A conventional flywheel can be used but an alternative comprises three concentric parts. The inner part is splined to shaft 43. The outer part extends to the flywheel's rim. Between them is a tough, slightly elastic part that absorbs some of the initial ignition jolt.~~

First, consider the benefit of overlapping power pistons on the power stroke controlled by the engine computer 7 (FIG 5). For example, a 2-stroke, 6 cyl engine with a 9" piston stroke would have the 1^{st} piston 6" after tdc, the 2^{nd} piston 3" after tdc and the 3^{rd} piston igniting at tdc. The 6 pistons continuously cycle through their power strokes in that sequence. The power added by the 3^{rd} piston is reduced by the combined remaining power of the 1^{st} and 2^{nd} pistons resulting in fuel savings and smooth power shaft rotation.

Underlying Mathematics.

Definitions

1 BTU = 778 ft-lbf.

1 hp = 550 ft-lbf/sec.

$2\pi r'$ = length of 1-way clutch rim at connecting rod contact. (ft)

bore – cylinder diameter. (in.)

Cp – cylinder pressure calculated from known bore size. (psi)

Dp – displacement. (cu.in.)

E – engine efficiency.

F – combustion force per piston. (lbf)

F' – estimated combustion pressure per piston. (psi) Used to find the bore size. (in.)

Fg – fuel flow rate. (gals/hr)

Fr – fuel flow rate. (lbm/sec)

Fw – fuel's weight. (lbm/gal.)

hp – shaft horsepower.

k = 2 or 4 (k = 2 for a 2-stroke, k = 4 for a 4-stroke.)

Lo = total efficiency – engine efficiency. (0.0 < Lo < 1.0)

n – number of active pistons. 2,4,6,8, ...

n/k – number of overlapping pistons simultaneously cycling through the power stroke.

Ps – length of piston's stroke. (in.)

Qc – fuel's energy density. (BTU/lbm)

r – radius of cylinder. (in)

r' – 1-way clutch radius at connecting rod contact. (ft)

Rv – power shaft's rotation rate. (rpm)

T – torque per piston. (lbf-ft)

T' – total shaft torque. (lbf-ft)

Vp – piston velocity. (ft/sec)

Equations:

$Vp = \pi(r')(Rv)/(30)$ Piston rod's and the 1-way clutch's rim speeds are equal at contact.

$r' = 30(Vp)/\pi(Rv)$ **r',Vp, Rv** are central to this engine's design and operation.

$Rv = 30(Vp)/(\pi r')$

$F = 550hp(k)/(nVp)$

$T = F(r')$

$T' = nT/k$

$F' = F/[\pi(r^2)]$

$r^2 = F/(\pi F')$

$bore = 2[F/(\pi F')]^{.5}$

$F = \pi F'(bore^2)/4$

$Cp = 4F/(\pi bore^2)$

$Dp = \pi(Ps)(n)(bore/2)^2$

$Fr = 550hp/[778(Qc)(1-Lo)]$

$Lo = 1 - 550hp/778(Qc)(Fr)$

$E = 1 - Lo$

$E = 550hp/778(Qc)(Fr)$

$Fg = Fr(3600)/(Fw)$

The examples next are only to illustrate how the underlying mathematics can be used to find basic engine specifications. Input values are estimates.

Examples:

1. A 2 cylinder, 2-stroke, 45 hp engine.

Let: **hp** = 20; **k** =2; **n** = 2; **F′** = 575 psi; **r′** = .5 ft; **F** = 1100 lbf; **Ps** = 4.5 in;

Fw = 6 lb/gal; **Qc** = 20500; **Lo** = .35

Vp = 10 ft/sec = $550(20)(2)/(2)(1100)$

bore = 1.561 in. = $2(1100/575\pi)^{.5}$ Set bore size at most frequently used 20 hp.

Rv = 191 rpm. = $30(10)/(.5\pi)$ Reduction gear may be required.

Dp = 17.2 cu.in. = $\pi(1.561/2)^2(4.5)(2)$

Fr = .001061074 lbm/sec. = $550(20)/(778)(20500)(1-.35)$

Fg = .6367 gals/hr. = $.001061074(3600)/6$

T = 550 lbf-ft. = 1100(.5)

Let: **hp** = 45; **k** =2; **n** = 2; **F′** = 575 psi; **r′** = .5 ft; **Vp** = 15 ft/sec.

bore = 1.561 in. = $2(1100/575\pi)^{.5}$ bore size same as 20 hp.

F = 1650 lbf. = 550(45)(2)/(2)(15)

Rv = 286 rpm. = $30(15)/(.5\pi)$ Reduction gear probably required.

Cp = 862 psi. = $4(1650)/\pi(1.561^2)$

Fr = .002387418 lbm/sec. = 550(45)/(778)(20500)(1-.35)

Fg = 1.432 gals/hr. = . 002387418(3600)/6

T = 825 lbf-ft. = 1650(.5)

2. A 4 cylinder, 4-stroke 400 hp engine.

Let: **hp** = 50; **Vp** = 14 ft/sec; **F′** = 700 psi; **n** = 4; **k** = 4; **r′** = .75 ft = 9 in.; **Ps** = 5.0 in;

Fw = 6 lb/gal; **Qc** = 20500; **Lo** = .35

F = 1964 lbf. = 4(50)(550)/4(14)

Rv = 178 rpm = $30(14)/(.75\pi)$

bore = 1.890 in. = $2(1964/700\pi)^{.5}$ Set bore size at most frequently used 50 hp.

Dp = 56 cu.in. = $\pi(1.890/2)^2(5.0)(4)$

Fr = .002652686 lbm/sec. = 550(50)/(778)(20500)(1-.35)

Fg = 1.592 gals/hr. = .002652686(3600)/6

T = T′ = 1473 lbf-ft. = 1964(.75)

Let: **hp** = 400; **Vp** = 27 ft/sec; **F′** = 800 psi; **n** = 4; **k** = 4; **r′** = .75 ft = 9 in.

F = 8148 lbf. = 4(400)(550)/4(27)

Rv = 344 rpm = $30(27)/(.75\pi)$

bore = 1.890 in. bore size same as 50 hp.

Cp = 2904 psi. = $4(8148)/\pi(1.890^2)$

Fr = .02122149 lbm/sec. = 550(400)/(778)(20500)(1-.35)

Fg = 12.733 gals/hr. = . 02122149(3600)/6

T = T′ = 6111 lbf-ft. = 8148(.75)

* If this engine were a 2-Stroke, there could be 50% power stroke overlap with both pairs active.

At low power, a pair of pistons could be stopped without load on the engine.

3. A 6 cylinder, 2-stroke 1200 hp. engine.

Let: **hp** = 700; **Vp** = 25 ft/sec; **F′** = 800 psi; **n** = 6; **k** = 2; **r′** = .75 ft = 9 in; **Ps** = 6.0 in.

F = 5133 lbf. = 2(700)(550)/6(25)

Rv = 318 rpm = $30(25)/(.75\pi)$

bore = 2.858 in. = $2(5133/800\pi)^{.5}$ Set bore size to most frequently used power.

Dp = 231 cu.in. = $\pi(2.858/2)^2(6.0)(6)$

T = 3850 lbf-ft. = 5133(.75)

T′ = 1150 lbf-ft. = 6(3850)/(2)

Let: **hp** = 1200; **Vp** = 35 ft/sec; **n** = 6; **k** = 2; **r′** = .75 ft. = 9 in.

F = 6285 lbf. = 2(1200)(550)/[(6)(35)]

Rv = 446 rpm = $30(35)/(.75\pi)$

bore = 2.858 in. bore size same as 700 hp.

Cp = 980 psi. = $4(6285)/\pi(2.858^2)$

T = 4714 lbf-ft. = 6285(.75)

T′ = 1414 lbf-ft. = 6(4714)/2

4. An 8 cylinder (2 banks of 4 cyls. each), 4-stroke, 1200 hp engine.

Let: **hp** = 1200; **F′** = 800 psi; **n** = 8; **k** = 4; **Vp** = 35 ft/sec; **r′** = 1.25 ft; **Ps** = 6.0 in.

1 cyl. per 1-way clutch requiring eight 1-way clutches. 50% overlap.

Rv = 267 rpm = $30(35)/(1.25\pi)$

F = 9429 lbf. = 4(550)(1200)/[(8)(35)]

bore = 3.874 in. = $2(9429/800\pi)^{.5}$

Dp = 566 cu.in. = $\pi(3.874/2)^2(6)(8)$

T = 11786 lbf-ft. = 9429(1.25)

T′ = 23573 lbf-ft. = 8(11786)/4

5. A large 8 cylinder, 2-stroke, 8,000 hp marine engine.

Let: **hp** = 8000; **F′** = 800 psi; **n** = 8; **k** = 2; **Vp** = 28 ft/sec; **Rv** = 100 rpm; **Ps** = 10 in; **Lo** = .35;

Fw = 7.1 lbm/gal. (1 cyl./1-way clutch uses 8 1-way clutches. 75% power stroke overlap.)

F = 39286 lbf. = 2(550)(8000)/[(8)(28)]

r′ = 2.673 ft. = $30(28)/(100\pi)$ Units 89 (FIGs 7,8) are carried by a short rimmed outer race 5

with a single spoke 35 to reduce inertia.

bore = 7.907 in. = $2[(39286/800\pi)]^{.5}$ Set bore size to most frequently used power.

Dp = 3929 cu.in. = $\pi(7.907/2)^2(10)(8)$

Fr = .457600229 lbm/sec. = 550(8000)/(778)(19014)(1-.35)

Fg = 232.02 gals/hr. = .457600229(3600)/7.1

T = 105011 lbf-ft. = 39286(2.673)

T' = 420046 lbf-ft. = 8(105011)/2

* Reset power by activating/deactivating piston pairs then vary power by varying the fuel charge.

Discussion.

A pair of combustion cylinders 33 and related pairs of parts that include a pair of 1-way clutches (FIGs 1-3) make the basic 2-stroke engine in this invention. The clutch's inner race 4 is keyed to the power shaft 8. The outer race 5 carries a sector gear 12. Each gear 12 engages an opposite side of idler 40 whereby synchronous reverse motion is transmitted between the power piston 38 and the second piston 38 in the pair as the inner race 4 transmits the power to the shaft 8. Moving parts that are not shown with arrows 42 are presumed obvious.

Combining two pairs with idler 40A creates a 4-stroke shown in FIG 6 that will be described later under Interchanging 4-stroke and 2-stroke.

FIG 2 shows a gear mesh to transmit the piston's power between piston rod 18 and the outer race 5 of the 1-way clutch. Rod 18 reciprocates along a straight path 42. FIG 2 also shows a reciprocating starter 46 gear meshed with the outer race 5. By shifting race 5, the starter shifts both pistons 38 until ignition. Alternatively, shaft 43 can be used to shift the pistons until ignition. The 4-stroke version in FIG 6 needs one starter 46 (not shown).

One end of a V-belt or a chain 9 is fastened to the outer race 5 (FIGs 1,3). The way it is wrapped around race 5 always keeps it taut, which prevents backlash as it rotates race 5 in response to the power stroke. Rod 18 is connected to the other end of the belt or chain 9 with a suitable fastener 41.

The 1-way clutch's override feature in this engine allows output shaft 8 and the clutch's inner race 4 to rotate independently of the pistons 38 when the inner race's speed is greater than the outer race 5 speed. This feature creates regenerated energy that is collected in an energy storage device 26 (FIG 5) available, e.g. for dumping to shaft 8 on demand or generating electricity.

The fixed length torque arm 10 (FIGs 2,3) causes instant peak torque at the beginning of the power stroke. A connecting rod guide 21, secured to housing 15, eliminates side thrust and reduces wear by keeping the piston 38 square in its cylinder. Wrist pins and piston skirts are not needed. The guide 21 is combined with a decelerator mechanism (FIG 4) to stop piston 38 at or near top dead

center. The decelerator includes a node 19 that is part of each rod 18 in a pair and a spring 45 for each node. The spring is encased in the guide 21. An opening in the housing 15 allows easy replacement of the spring. The spring absorbs the impact of node 19 to halt the motion of piston 38, which is then accelerated on its power stroke by timely expanding combustion gases. The impact is reduced because node 19 is decelerating due to the power loss of the power piston to the shaft 8. The decelerator is positioned to prevent backlash of the gears 12 (FIGs 1,6) that mesh with idler 40.

A computer 7 (FIG 5) monitors input from the throttle 6 and shaft power from the sensor 22 on shaft 8 through leads 23 to determine the size of the combustion charge to transmit to the cylinders through injector lines 24. The position of piston 38 is monitored through sensors 22 on shaft 43 and used for ignition timing. By monitoring the motion of each shaft 43 in several pairs, the computer controls timing between the unconnected pairs in a 2-stroke embodiment. The computer begins a power stroke with a piston in one pair when a piston in another pair is partly through its power stroke. In a 2-stroke, 50% power stroke overlap and smooth rotation of the shaft 8 is had with two unconnected pairs (four cylinders). Greater overlap is gained with more pairs.

Small Flywheels.

A small, suitable flywheel 48 is splined to the end of shaft 43 (FIG 3) to briefly increase chamber pressure for a more complete combustion with decreased emissions. Then, it dispenses the regenerated energy that it gains to moderate the speed of the pistons 38. A conventional flywheel can be used but an alternative comprises three concentric parts. The inner part is splined to shaft 43. The outer part extends to the flywheel's rim. Between them is a tough, slightly elastic part that absorbs some of the initial ignition jolt.

An equivalent to the flywheel 48 (not shown) is to construct the inner race 4 with springs like the flywheel carried behind the engine of conventional vehicles. The inner race absorbs the ignition jolt.

Re: Inventor: Robert Louis Giuliani
Application no. 10/643274
Filing date: 08/18/2003
Title: Interchangeable 2-stroke or 4-stroke High Torque Power Engine
CIP of application no.10/252,927 filing date 09/24/2002
Art Unit: 3748
Confirmation no. 4067

REMARKS

1. The part to be deleted in the above application no.10/643274 by this 2nd amendment is shown with strikethroughs on sheets 2-5 which includes the 1st amendment and also part of the original CIP that was not amended with the 1st amendment. The strikethroughs go through all lines between the section heading DETAILED DESCRIPTION OF THE INVENTION and the subsection heading **Interchanging 4-Stroke and 2-Stroke** of the original CIP. It is to be entirely replaced with this 2nd amendment which includes all the underlined on sheets 6-11.

2. There is one added paragraph that consists of two sentences. It is the last paragraph on sheet 11 and it is not underlined in agreement with "Revised Amendment Practice: effective July 30, 2003".

3. The claims are on sheets 12-14. There is one amendment to Claim 7. The Claim 7 amendment with new parts underlined is shown on sheet 12.

4. There are no amendments to the drawings.

If there are questions, I can be contacted by email or phone number. If by phone, the best time to call is 0730-0830 Hawaii time.

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23 MAR 2004

Mail Stop Non-fee Amendment
Commissioner for Patents
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Re: Inventor: Robert Louis Giuliani
Application no. 10/643274
Filing date: 08/18/2003
Title: Interchangeable 2-stroke or 4-stroke High Torque Power Engine
CIP of application no.10/252,927 filing date 09/24/2002
Art Unit: 3748
Confirmation no. 4067

INTRODUCTORY COMMENTS

This is the 3rd amendment to this CIP application no. 10/643274. The 1st amendment was dated 27 NOV 2003. The 2nd amendment was dated 27 FEB 2004. All the claims in the 2nd amendment are included in this 3rd amendment with Claim 20.(withdrawn) on sheet 4 and replaced by claim 21.(new).

Claim7 was changed in the 2nd amendment so its parenthetical status is shown as (previously presented) in this 3rd amendment.

There are no amendments to the drawings or specifications in this 3rd amendment.

See the REMARKS on sheets 5 and 6.

I believe I am following the correct procedure laid out in the Revised Amendment Practice – Effective Date: July 30, 2003. If there are questions, I can be contacted by email or telephone. If by phone, the best time to call is 0730-0830 Hawaii time, 5 hours later than East Coast time.

RLGiuliani
R.L. GIULIANI
Inventor/Applicant

REMARKS

Claim 21 differs from existing technology by increasing the length **L** of the piston rod 18 while keeping the length **r** of the crank arm fixed (FIG 14). The trend in engines is to have a short stroke and a short piston rod to gain high rpm. "This means that an engine can run much faster - up to 15,000 rpm in a Champ Car engine – but relatively little torque." See attached copy from howstuffworks taken from the internet.

A copy of 1 page of "The Most Powerful Diesel Engine in the World" is attached. It shows the "crosshead" design that keeps the piston square in the cylinder but uses a short connecting rod. This is opposite to my long connecting rod 18 with a small angle θ.

Claim 21.(new) is justified **entirely** by the math shown in the BACKGROUND OF THE INVENTION section of the CIP pages 2,3. The math shows how a longer length **L** of the piston rod 18 with a fixed crank arm **r** allows a small displacement of the piston 38 and small angle θ to cause a large angular displacement of the shaft 8 **before** reaching the $\lim_{\theta \to 0.0} \cos\theta = 1.0$.

Claim 21 intends that the offset piston description that dominates the CIP can be reached with increasing effectiveness by increasing the piston rod length **L** while keeping the length of the crank arm **r** fixed. Claim 21 does not intend to reduce the importance of the offset piston emphasized and claimed in the CIP. The offset piston explains several important features that cannot be achieved with the engine claimed by claim 21.

Equations from the CIP are used in the following 2 examples to make the point for Claim 21. The large increase in α with a small displacement of piston 38 and small increase in θ is evident:

$\cos\theta = \cos\{\sin^{-1}[(r/L)\sin\alpha]\}$ taken from the CIP page 3.

$\cos\Phi = \cos(90 - \{\alpha + \sin^{-1}[(r/L)\sin\alpha]\})$ taken from the CIP page 3.

$a = r(1 - \cos\alpha)$ taken from the CIP page 2.

Solve Cos θ for α:

$\theta = \sin^{-1}[(r/L)\sin\alpha]$

$\sin\theta = (r/L)\sin\alpha$

$\sin\alpha = (L/r)\sin\theta$

$\alpha = \sin^{-1}[(L/r)\sin\theta]$

Example 1

Let: $\theta = 17°$, $L = 5"$, $r = 1.5"$

$\alpha = \sin^{-1}[(10/1.5)\sin 17°] = 77.052°$ FVI is tangent to crank circle **d.**

$\cos\theta = \cos 17° = .9563$

$\cos\Phi = \cos(90 - \{77.052 + \sin^{-1}[(1.5/10)\sin 77.052)]\} = .9969$

$\cos\theta \cos\Phi = (.9563)(.9969) = .9533 = 95.33\%$ efficiency when FV1 is tangent to circle **d.**

$a = 1.5"(1 - \cos 77.052°) = 1.164"$ piston displacement when FV1 is tangent to circle **d.**

Example 2

Let: $\theta = 8.5°$, $L = 10"$, $r = 1.5"$

$\alpha = \sin^{-1}[(10/1.5)\sin 8.5°] = 80.196°$ angular displacement when FV1 is tangent to circle **d.**

$\text{Cos}\ \theta = \text{Cos}\ 8.5° = .9890$

$\text{Cos}\ \Phi = \text{Cos}(90 - \{80.196 + \text{Sin}^{-1}[(1.5/10)\text{Sin}\ 80.196)]\} = .9997$

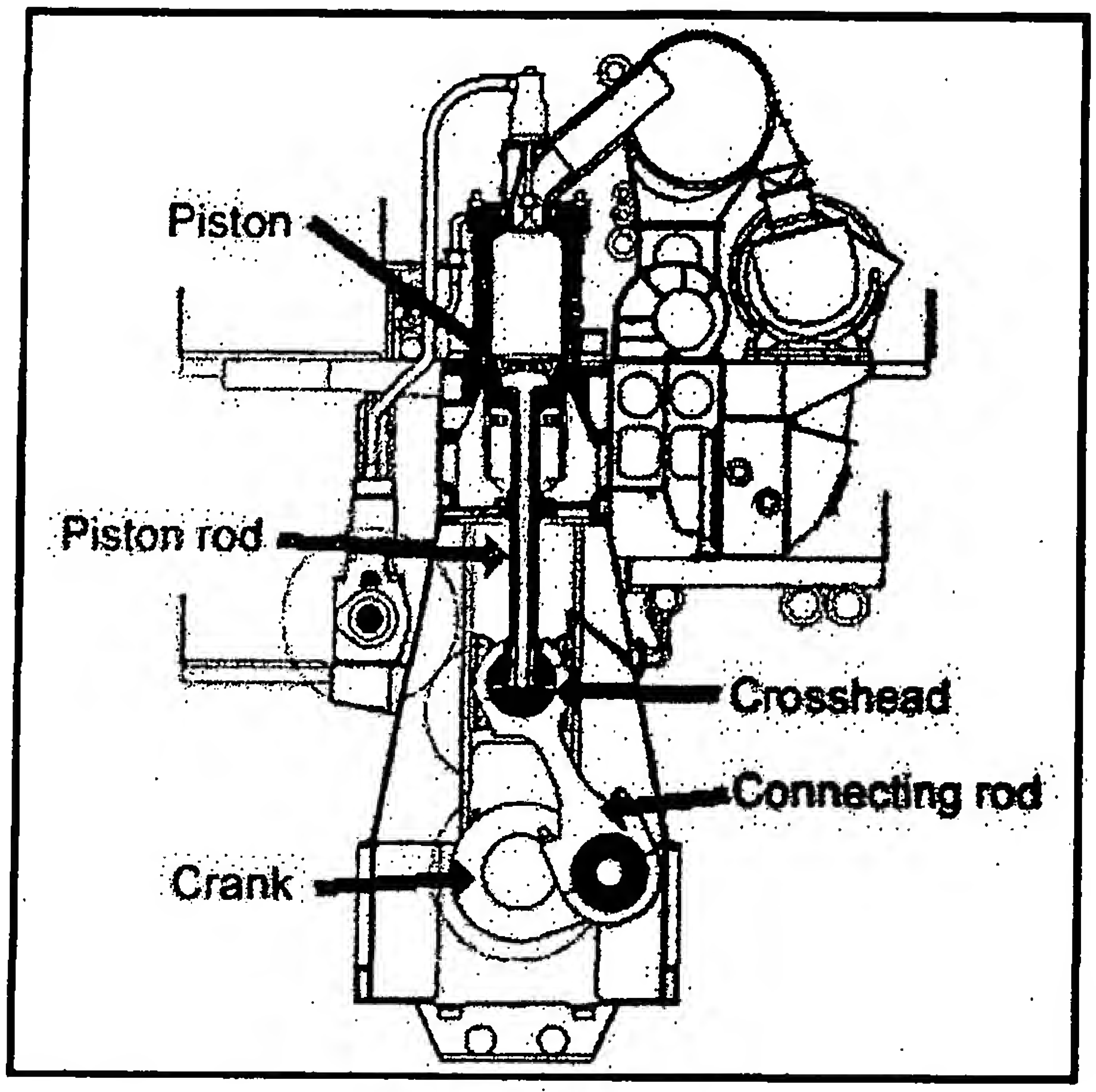
$\text{Cos}\ \theta\ \text{Cos}\ \Phi = (.9890)(.9997) = .9887 = 98.87\%$ efficiency when FV1 is tangent to circle **d.**

a = 1.5"(1 – Cos 80.196°) = 1.245" piston displacement when FV1 is tangent to circle **d.**

The crank engine's efficiency is still at or near zero at or near top dead center.

R.E. Giuliani

Inventor/Applicant



The internals of this engine are a bit different than most automotive engines.
The top of the connecting rod is not attached directly to the piston. The top of the connec attaches to a "crosshead" which rides in guide channels. A long piston rod then connects th crosshead to the piston.
I assume this is done so the the sideways forces produced by the connecting rod are abs the crosshead and not by the piston. Those sideways forces are what makes the cylinders i engine get oval-shaped over time.

Installing the "thin-shell" bearings. Crank & rod journals are 38" in diameter and 16"

Why do big diesel engines and race car en have such different horsepower ratings?

Let's start by answering the question and then look at why the world works tha

The answer to your question has to do with the way the two engines are desig liter diesel engine has a long **stroke**. That means that the piston is traveling a distance up and down in its cylinder on each cycle. A racing engine, on the otr a short stroke. The piston in a racing engine has a large diameter for the engir goes up and down a relatively short distance on each cycle. This means that a engine can run much faster -- up to 15,000 RPM in a Champ Car engine -- but relatively little torque. A large diesel engine usually cannot get above 2,000 RF huge torque because of the long stroke. The torque is what lets your engine pı load up a hill.

So why does an engine with huge torque and low maximum RPM get a low ho rating? If you have read the article entitled How Horsepower Works, then you I horsepower is equal to 33,000 foot-pounds of work per minute. By this measuı can raise 33 pounds 1,000 vertical feet in a minute, or 330 pounds 100 feet in 3,300 pounds 10 feet in a minute, and so on.

What an engine naturally produces, however, is **torque**. Think about one pisto gasoline engine. When the gasoline ignites, it pushes on the piston, and the pi pressure on the crankshaft, causing it to turn. The crankshaft feels some numt pounds of torque in the process. There are three variables that affect torque:

- The size of the piston face
- The amount of pressure that the ignited fuel applies to the face of the pi:
- The distance the piston travels on each stroke (therefore the diameter o crankshaft). The bigger the diameter of the crankshaft, the bigger the le' therefore the more torque.

There is a direct relationship between horsepower and torque. You can convei horsepower with the following equation:

HP = Torque * RPM / 5,252

That 5,252 number, by the way, comes from dividing 33,000 by (2 * pi). Imagir 33,000 foot-pounds and walking it around in a circle rather than a straight line. if you took a 10 foot pole and attached it to a vertical axle, the circumference o

circumference = 10 * 2 * pi = 62.8 feet

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If one horse is pushing on the pole with 100 pounds of force (1,000 foot-pound
it can move the pole at 5.25 RPM. Torque and horsepower are directly related
other.

You can see from the horsepower equation that high RPM values favor horsep
take an engine with a certain torque and run it at very high revs, it can generat
horsepower even though its torque hasn't changed at all. A racing engine can
relatively low torque, but because it can rev so high it gets a great horsepower
diesel has huge torque, but "gets no respect" in terms of horsepower because
ever get above 2,000 RPM. This "makes sense" -- if two engines produce the
the one that can do it more times per minute does more work and therefore ha
power.

Search Google

Why+do+big+diesel+e

powered by Google Go

The difference in maximum RPM ratings also tells you why trucks need so ma
race car engine might idle at 1,000 RPM and can accelerate to 15,000 RPM --
of 15. A big diesel might have a multiplier of only 2 or 3. Because the RPM ran
minimum and maximum is so small on a diesel, there needs to be lots of differ
keep the engine in its productive RPM range at any speed.

Here are several interesting links:

- How Champ Cars Work
- How NASCAR Race Cars Work
- How Diesel Engines Work
- How Horsepower Works
- Glossary of CART terms (see "Engine")

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30 MAR 2004

Mail Stop Non-fee Amendment
Commissioner for Patents
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Alexandria, VA 22313-1450

Re: Inventor: Robert Louis Giuliani
Application no. 10/643274
File date: 08/18/2003
Title: Interchangeable 2-stroke or 4-stroke High Torque Power Engine
CIP of application no.10/252,927 file date 09/24/2002
Art Unit: 3748
Confirmation no. 4067

INTRODUCTORY COMMENTS

This is the 4th amendment to this CIP application no. 10/643274. The 1st amendment was dated 27 NOV 2003. The 2nd amendment was dated 27 FEB 2004. The 3rd amendment was dated 23 MAR 2004.

There is one amendment to Claim 7 in this 4th amendment. It is the 3rd amendment to Claim 7. No other claims are affected.

One entire subsection in the specification has been replaced including its title.

One drawing (FIG 4) has been changed.

See the REMARKS on sheet 7.

This amendment is believed to be in agreement with Revised Amendment Practice – Effective Date: July 30, 2003.

I can be contacted by the above email or telephone. If by phone, the best time to call is 0730-0830 Hawaii time, 6 hours later than East Coast daylight saving time.

RL Giuliani
R.L. GIULIANI
Inventor/Applicant

~~**Small Flywheels.**~~

~~A small, suitable flywheel 48 is splined to the end of shaft 43 (FIG 3) to briefly increase chamber pressure for a more complete combustion with decreased emissions. Then, it dispenses the regenerated energy that it gains to moderate the speed of the pistons 38. A conventional flywheel can be used but an alternative comprises three concentric parts. The inner part is splined to shaft 43. The outer part extends to the flywheel's rim. Between them is a tough, slightly elastic part that absorbs some of the initial ignition jolt.~~

~~An equivalent to the flywheel 48 (not shown) is to construct the inner race 4 with springs like the flywheel carried behind the engine of conventional vehicles. The inner race absorbs the ignition jolt.~~

Moderated Combustion Pressure.

There are least three ways to absorb excessive peak cylinder pressure and dispense it back to the chamber 33 so that a moderated pressure is maintained during the piston stroke to achieve a better burn which increases efficiency and reduces pollution and waste heat.

The first way (FIG 4) uses a two-part piston rod 18 and 18A with a spring 16 between the two parts. Spring 16 is connected to the two parts such that its compression and expansion are not affected. Part 18 has an extension 14 that extends through the center of spring 16 into a cylinder 13 in part 18A (shown in cross section) to keep the spring 16 centered on the axis of the two piston rod parts. Side thrust, if any, will be negligible on the parts because of the combined guide 21 and the piston 38 being square in the cylinder 33.

There are two channels 2 on opposite sides of the cylinder 13 that are aligned with the axis of the cylinder. A small projection 3 on the extension 14 reaches into each channel to prevent angular motion of part 18A and piston 38.

A second way includes a small, suitable flywheel 48 splined to the end of shaft 43 (FIG 3). A conventional flywheel can be used but an alternative comprises three concentric parts. The inner part is splined to shaft 43. The outer part extends to the flywheel's rim. Between them is a tough, slightly elastic part that absorbs some of the initial ignition jolt.

A third way is to construct the inner race 4 with springs like the flywheel carried behind the engine of conventional vehicles. The inner race performs like the flywheel.

REMARKS

This 4th amendment changes the FIG 4 drawing in the above CIP. Because FIG 4 and FIG 5 are on the same sheet in the CIP, I have included both in their original positions on the enclosed unnumbered drawing sheet following this sheet 7. There is no change to FIG 5.

The changed FIG 4 requires revising the subsection in the 2nd amendment titled, "**Small Flywheels**". The entire subsection including the title is shown with srikethroughs on sheet 2. The entire revised subsection is shown underlined on sheet 3 with the new title, "**Moderated Combustion Pressure.**". Delete the entire subsection shown on sheet 2, including its title, and replace it with the entire revised subsection, including the new title, shown on sheet 3 of this 4th amendment.

The changed FIG 4 also requires a third change to Claim 7. This 4th amendment does not change any other claims. Claim 7 was changed in the 2nd amendment and again in the 3rd amendment so its parenthetical status is shown as (previously presented) in the 3rd amendment. The changed Claim 7 in this 4th amendment has the parenthetical status (thrice amended). (thrice amended) is used on advice from the Inventor's Assistance Ctr. who I called for advice after the 2nd amendment was mailed to the PTO. The complete listing of all the claims are on sheets 4-6.

Claim 20 was withdrawn in the 3rd amendment and replaced by the new claim 21. I am uncertain which parenthetical expression(s) apply to them in this 4th amendment so I copied them intact into this 4th amendment. They are on sheet 6

The matter in this 4th amendment is equivalent to existing matter. There is no new matter in this amendment.

R.L. Giuliani

R.L. Giuliani
Inventor/Applicant

REPLACEMENT SHEET

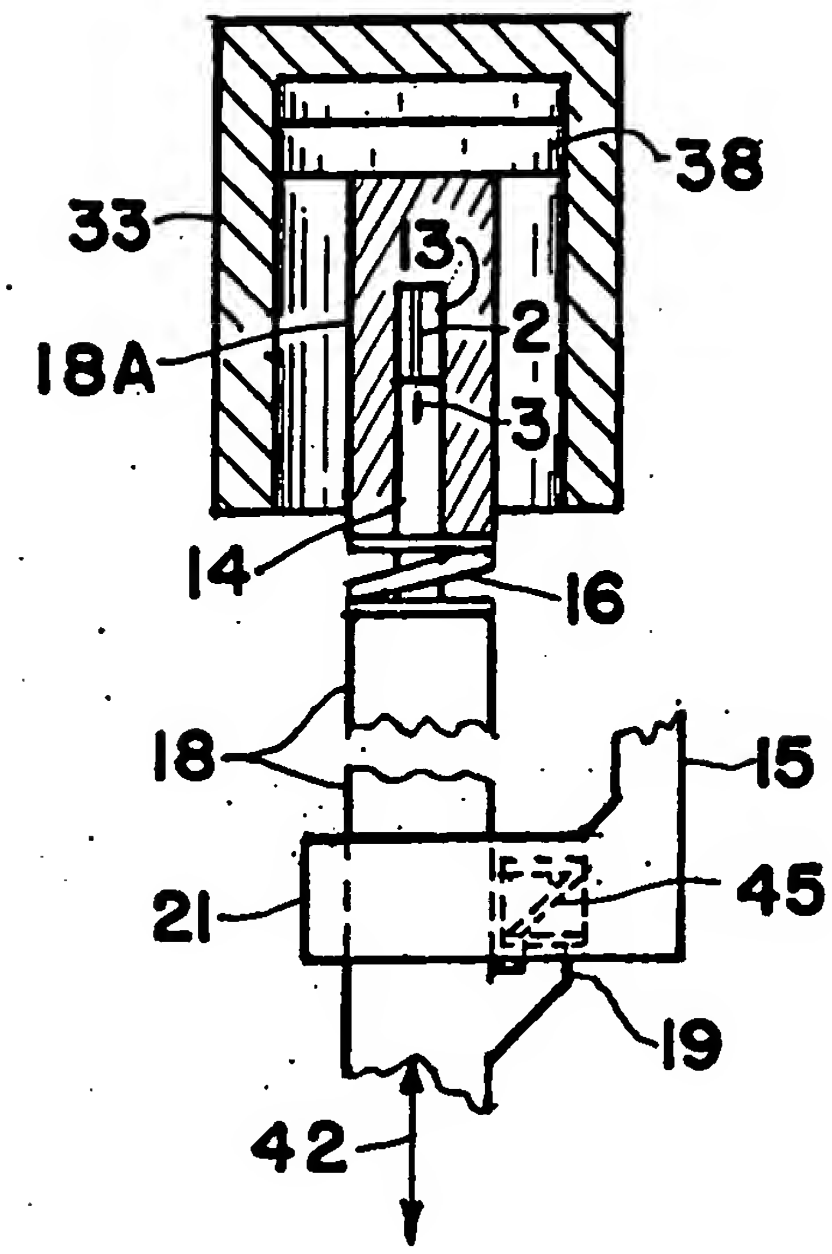

FIG. 4

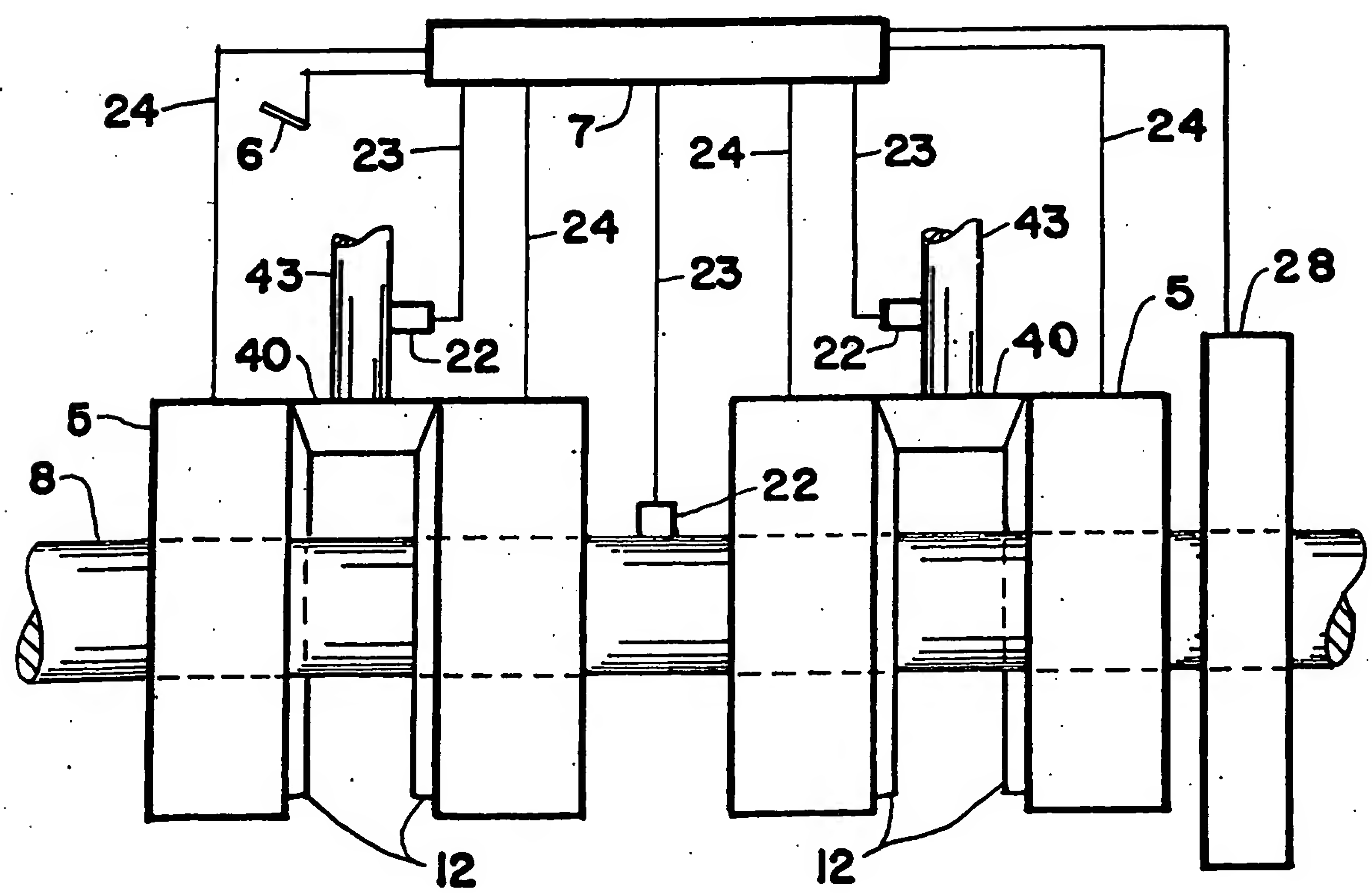

FIG. 5

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5 APR 2004

Mail Stop Non-fee Amendment
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Re: Inventor: Robert Louis Giuliani
Application no. 10/643274
File date: 08/18/2003
Title: Interchangeable 2-stroke or 4-stroke High Torque Power Engine
CIP of application no.10/252,927 file date 09/24/2002
Art Unit: 3748
Confirmation no. 4067

INTRODUCTORY COMMENTS

This is the 5th amendment to this CIP application no. 10/643274. The 1st amendment was dated 27 NOV 2003. The 2nd amendment was dated 27 FEB 2004. The 3rd amendment was dated 23 MAR 2004. The 4th amendment was dated 30 MAR 2004.

This 5th amendment is to: (1) correct a fatal flaw in the specification that fails to properly describe the intake stroke in the 4-Stroke embodiment of this engine, (2) include a version of the 1-way clutch in the written specification that references the changed FIG 12 on the last sheet and (3) revise the **Underlying Mathematics** subsection to make it easier to understand by reducing 5 examples to 1 and more comprehensive with additional equations.

See the REMARKS on sheet 13.

This amendment is believed to be in agreement with Revised Amendment Practice – Effective Date: July 30, 2003. Hopefully, it will be the last for this application.

I can be contacted by the above email or telephone. If by phone, the best time to call is 0730-0830 Hawaii time, 6 hours later than East Coast daylight saving time.

R.L. GIULIANI
Inventor/Applicant

Underlying Mathematics.

Definitions:

1 BTU = 778 ft-lbf
1 hp = 550 ft-lbf/sec.
$2\pi r'$ = length of 1-way clutch rim at connecting rod contact. (ft)
bore – cylinder diameter. (in.)
Cp – cylinder pressure calculated from known bore size. (psi)
Dp – displacement (cu.in.)
E – fuel efficiency
F – combustion force per piston. (lbf)
Fg – fuel flow rate (gals/hr)
Fi – force on the inner race (lbf)
Fr – fuel flow rate (lbm/sec)
Fu – force per unit 89 (lbf) See FIGs 7 or FIG 8 for unit 89.
Fw – fuel's weight (lbm/gal.)
hp – shaft horsepower.
k = 2 or 4 (k = 2 for a 2-stroke. k = 4 for a 4-stroke.)
Lo – ~~power losses~~ **[[( ]]**fraction of power lost ~~fuel's energy density).~~
n – number of active pistons. 2,4,6,8, …
n/k – number of overlapping pistons cycling through the power stroke.
Nu – number of units 89 (FIGs 7,8).
[[F']] **Pp** – estimated combustion pressure per piston. (psi) Used to find the bore size. (in.)
Ps – length of piston's stroke. (in.)
Qc – fuel's energy density. (BTU/lbm)
r – radius of cylinder. (in)
r' – 1-way clutch radius at connecting rod contact. (ft)
ri – radius of the 1-way clutch inner race. (ft)
Rv – power shaft's rotation rate. (rpm)
Sp – Center to center spacing between units 89 (FIGs 7,8). (ft)
T – torque per piston. (lbf-ft)
T' – total shaft torque. (lbf-ft)
Vp – piston velocity. (ft/sec)

Equations:

$Vp = \pi(r')(Rv)/(30)$ Piston rod's and the 1-way clutch's rim speeds are equal at contact.
$r' = 30(Vp)/\pi(Rv)$ **r',Vp, Rv** are central to this engine's design and operation.
$Rv = 30(Vp)/(\pi r')$
$F = 550hp(k)/(nVp)$
$\cancel{F = 16500(hp)(k)/\pi(n)(Rv)(r')}$
$hp = F(n)(Vp)/550$
$hp = Fr[778(Qc)(1-Lo)]/550$
$T = F(r')$
$T' = nT/k$
[[F']]**Pp** $= F/[\pi(r^2)]$

$r^2 = F/\pi[[F']]\underline{Pp})$

$bore = 2[F/(\pi[[F']]\underline{Pp})]^{.5}$

$F = \pi[[F']]\underline{(Pp)}(bore^2)/4$

~~Sp = 550hp(1+lo)~~

~~Fr = (Sp)/778Qe)~~

~~Fr = (F)(n)(Vp)/[k(778Qe)]~~

$\underline{Fi = F(r')/ri}$

$\underline{Nu = 2\pi(ri)/Sp}$

$\underline{Fu = F(r')(Sp)/2\pi(ri^2)}$

$\underline{Fu = F(r')/(ri)(Nu)}$

$\underline{Fu = Fi/Nu}$

$\underline{Cp = 4F/(\pi bore^2)}$

$\underline{Dp = \pi(bore/2)^2(Ps)(n)}$

$\underline{Fr = 550hp/[778(Qc)(1-Lo)]}$

$\underline{Lo = 1 - 550hp/778(Qc)Fr}$

$\underline{E = 1 - Lo}$

$\underline{E = 550hp/778(Qc)Fr}$

$\underline{Fg = Fr(3600)/(Fw)}$

The following example demonstrates the effectiveness of the Underlying Mathematics in finding the correct general engine specifications from which the rest of the engine can be built. The given values are hypothetical. This example is for a low power engine, e.g. lawn mowers and outboard marine, but the math can be applied to any size engine.

Example:

Given:

[[F']]<u>Pp</u> = 100 psi; **F** = 300 lbf; **Vp** = 3.5 ft/sec; **r'** = 4.5" = .375 ft; <u>**ri** = 3.75" = .3125 ft;</u>
k = 2; **n** = 2; **Qc** = 20500; **Lo** = .35; **Fw** = 6 lbm/gal; <u>**Sp** = 6" = .5ft</u>

hp = 300(2)(3.5)/[2(550)] = **1.909**

r^2 = 300/(100π) = **.9549** in^2

bore = 2[300/(100π)]$^{.5}$ = **1.9544** in.

Rv = 30(3.5)/(.375π) = **89.13** rpm

<u>**Fi** = 300(4.5)/3.8 = **355.37** lbf</u>

<u>**Nu** = 2π(3.8)/6 = **4**</u>

<u>**Fu** = 300(4.5)/[6(3.75)] = **60** lbf.</u>

T = 300(.375) = **112.5** lbf-ft

Fr = 550(1.909)/[778(20500)(1-.35)] = **.000101284** lbm/sec.

Fg = .000101284(3600)/6 = **.060770629** gals/hr.

Given: **hp** = 10; **F** = 380 lbf.

Vp = 550(10)(2)/(2)(380) = **14.5** ft/sec.

Rv = 30(14.5)/(.375π) = **369** rpm.

Fi = 380(4.5)/3.8 = **450** lbf

Fu = 380(4.5)/[6(3.75)] = **113** lbf.

Cp = 4(380)/[(1.9544^2)π] = **126.67** psi.

T = 380(.375) = **142.5** lbf-ft

Fr = 550(10)/[778(20500)(1-.35)] = **.000530537** lbm/sec.

Fg = .000530537(3600)/6 = **.318322345** gals/hr.

~~The examples next are only to illustrate how the underlying mathematics can be used to find basic engine specifications. Input values are estimates.~~

~~Examples:~~

~~1. A 2 cylinder, 2-stroke, 45 hp engine.~~

~~Let: **hp** = 20; **k** = 2; **n** = 2; **F′** = 575 psi; **r′** = .5 ft; **F** = 1100 lbf; **Ps** = 4.5 in;~~

~~**Fw** = 6 lb/gal; **Qe** = 20500; **Lo** = .35~~

~~**Vp** = **10 ft/sec** = 550(20)(2)/(2)(1100)~~

~~**bore** = **1.561 in.** = 2(1100/575π)$^{.5}$ Set bore size at most frequently used 20 hp.~~

~~**Rv** = **191 rpm.** = 30(10)/(.5π) Reduction gear may be required.~~

~~**Dp** = **17.2 cu.in.** = π(1.561/2)^2(4.5)(2)~~

~~**Fr** = **.001061074 lbm/sec.** = 550(20)/(778)(20500)(1-.35)~~

~~**Fg** = **.6367 gals/hr.** = .001061074(3600)/6~~

~~**T** = **550 lbf-ft.** = 1100(.5)~~

~~Let: **hp** = 45; **k** = 2; **n** = 2; **F′** = 575 psi; **r′** = .5 ft; **Vp** = 15 ft/sec.~~

~~**bore** = **1.561 in.** = 2(1100/575π)$^{.5}$ bore size same as 20 hp.~~

~~**F** = **1650 lbf.** = 550(45)(2)/(2)(15)~~

~~**Rv** = **286 rpm.** = 30(15)/(.5π) Reduction gear probably required.~~

~~**Cp** = **862 psi.** = 4(1650)/π(1.561^2)~~

~~**Fr** = **.002387418 lbm/sec.** = 550(45)/(778)(20500)(1-.35)~~

~~**Fg** = **1.432 gals/hr.** = .002387418(3600)/6~~

~~**T** = **825 lbf-ft.** = 1650(.5)~~

~~2. A 4 cylinder, 4-stroke 400 hp engine.~~

~~Let: **hp** = 50; **Vp** = 14 ft/sec; **F′** = 700 psi; **n** = 4; **k** = 4; **r′** = .75 ft = 9 in.; **Ps** = 5.0 in;~~

~~**Fw** = 6 lb/gal; **Qe** = 20500; **Lo** = .35~~

~~**F** = **1964 lbf.** = 4(50)(550)/4(14)~~

~~**Rv = 178 rpm** = $30(14)/(.75\pi)$~~

~~**bore = 1.890 in.** = $2(1964/700\pi)^{.5}$ Set bore size at most frequently used 50 hp.~~

~~**Dp = 56 cu.in.** = $\pi(1.890/2)^2(5.0)(4)$~~

~~**Fr = .002652686 lbm/sec.** = $550(50)/(778)(20500)(1-.35)$~~

~~**Fg = 1.592 gals/hr.** = $.002652686(3600)/6$~~

~~**T = T′ = 1473 lbf-ft.** = $1964(.75)$~~

~~Let: **hp** = 400; **Vp** = 27 ft/sec; **F′** = 800 psi; **n** = 4; **k** = 4; **r′** = .75 ft = 9 in.~~

~~**F = 8148 lbf.** = $4(400)(550)/4(27)$~~

~~**Rv = 344 rpm** = $30(27)/(.75\pi)$~~

~~**bore = 1.890 in.** bore size same as 50 hp.~~

~~**Cp = 2904 psi.** = $4(8148)/\pi(1.890^2)$~~

~~**Fr = .02122149 lbm/sec.** = $550(400)/(778)(20500)(1-.35)$~~

~~**Fg = 12.733 gals/hr.** = $.02122149(3600)/6$~~

~~**T = T′ = 6111 lbf-ft.** = $8148(.75)$~~

~~* If this engine were a 2-Stroke, there could be 50% power stroke overlap with both pairs active.~~

~~At low power, a pair of pistons could be stopped without load on the engine.~~

~~3. A 6 cylinder, 2-stroke 1200 hp. engine.~~

~~Let: **hp** = 700; **Vp** = 25 ft/sec; **F′** = 800 psi; **n** = 6; **k** = 2; **r′** = .75 ft = 9 in; **Ps** = 6.0 in.~~

~~**F = 5133 lbf.** = $2(700)(550)/6(25)$~~

~~**Rv = 318 rpm** = $30(25)/(.75\pi)$~~

~~**bore = 2.858 in.** = $2(5133/800\pi)^{.5}$ Set bore size to most frequently used power.~~

~~**Dp = 231 cu.in.** = $\pi(2.858/2)^2(6.0)(6)$~~

~~**T = 3850 lbf-ft.** = $5133(.75)$~~

~~**T′ = 1150 lbf-ft.** = $6(3850)/(2)$~~

~~Let: **hp** = 1200; **Vp** = 35 ft/sec; **n** = 6; **k** = 2; **r′** = .75 ft. = 9 in.~~

~~**F = 6285 lbf.** = $2(1200)(550)/[(6)(35)]$~~

~~**Rv = 446 rpm** = $30(35)/(.75\pi)$~~

~~**bore = 2.858 in.** bore size same as 700 hp.~~

~~**Cp = 980 psi.** = $4(6285)/\pi(2.858^2)$~~

~~**T = 4714 lbf-ft.** = $6285(.75)$~~

~~**T′ = 1414 lbf-ft.** = $6(4714)/2$~~

~~4. An 8 cylinder (2 banks of 4 cyls. each), 4-stroke, 1200 hp engine.~~

~~Let: **hp** = 1200; **F′** = 800 psi; **n** = 8; **k** = 4; **Vp** = 35 ft/sec; **r′** = 1.25 ft; **Ps** = 6.0 in.~~

~~1 cyl. per 1-way clutch requiring eight 1-way clutches. 50% overlap.~~

~~**Rv = 267 rpm** = 30(35)/(1.25π)~~

~~**F = 9429 lbf.** = 4(550)(1200)/[(8)(35)]~~

~~**bore = 3.874 in.** = 2(9429/800π)~~$^{.5}$

~~**Dp = 566 cu.in.** = π(3.874/2)~~^{2}~~(6)(8)~~

~~**T = 11786 lbf-ft.** = 9429(1.25)~~

~~**T′ = 23573 lbf-ft.** = 8(11786)/4~~

~~5. A large 8 cylinder, 2-stroke, 8,000 hp marine engine.~~

~~Let: **hp** = 8000; **F′** = 800 psi; **n** = 8; **k** = 2; **Vp** = 28 ft/sec; **Rv** = 100 rpm; **Ps** = 10 in; **Lo** = .35;~~

~~**Fw** = 7.1 lbm/gal. (1 cyl./1-way clutch uses 8 1-way clutches. 75% power stroke overlap.)~~

~~**F = 39286 lbf.** = 2(550)(8000)/[(8)(28)]~~

~~**r′ = 2.673 ft.** = 30(28)/(100π) Units 89 (FIGs 7,8) are carried by a short rimmed outer race 5~~

~~with a single spoke 35 to reduce inertia.~~

~~**bore = 7.907 in.** = 2[(39286/800π)]~~$^{.5}$ ~~Set bore size to most frequently used power.~~

~~**Dp = 3929 cu.in.** = π(7.907/2)~~^{2}~~(10)(8)~~

~~**Fr = .457600229 lbm/sec.** = 550(8000)/(778)(19014)(1-.35)~~

~~**Fg = 232.02 gals/hr.** = .457600229(3600)/7.1~~

~~**T = 105011 lbf-ft.** = 39286(2.673)~~

~~**T′ = 420046 lbf-ft.** = 8(105011)/2~~

~~* Reset power by activating/deactivating piston pairs then vary power by varying the fuel charge.~~

Interchanging 4-Stroke and 2-Stroke.

With reference first to FIG 3, the flexible chain 9 must be made stiff enough to pull the piston 38 down during the intake stroke in the 4-stroke engine. In this case, the outer race 5 is a cogwheel and the cogs fit between the chain links similar to a bicycle chain. Each side of each link has an extension that rides in an immovable channel that is secured to the engine. The two channels combine with the side extensions to prevent the chain from flexing out of mesh with the race 5 cogwheel during the intake stroke and without interfering with the other piston strokes. Both channels are shaped in an arc around the race 5 where they are connected with a solid cover over the chain to insure against the chain flexing. The cover ends near the position of fastener 41 in FIG 3 when piston 38 is at top dead center. The channels continue straight downward without the cover to prevent the chain from flexing when it is straight. The straight channels extend to a point slightly beyond the position of fastener 41 when the piston is at bottom dead center. The fastener is connected to the chain free of the extensions so that the channels do not interfere with the motion of the chain. This allows the fastener 41 to reach its highest point (FIG 3) where it is in position to begin the intake stroke and complete the stroke without interference from the channels or the cover.

~~There are at least two simple ways to effect this change.~~ There are at least two simple ways to change between a 2-stroke and a 4-stroke. In a 4-stroke, a sector gear 12 on two pairs engages idler 40A (FIG 6). A <u>removable</u> cap 54 having a hole is ~~removably secured, e.g.~~ threaded[[,]] to the engine 15. The shaft 43 of idler 40A has two diameters. The shorter one extends through the hole. A snap ring 56 on the shorter diameter abuts the cap and combines with the larger diameter that abuts the inside of the cap to prevent the idler 40A from axial movement which keeps the idler properly engaged with the two sector gears. When changing to a 4-stroke from a 2-stroke, the pistons must be correctly ~~aligned~~ <u>positioned</u> before engaging the idler with the sector gears. One of the correct ~~alignments~~ <u>positions</u> is shown in FIG 6 with 2 pistons at top dead center and 2 at bottom dead center. Power stroke overlap for a 4-stroke can be achieved by adding another bank of two pairs along the shaft 8 disengaged from the bank shown in FIG 6 or by adding separate pairs. ~~To avoid cluttering, FIGs 6,6A show the splined end of shafts 43 without flywheels 48.~~

The separation 1 in FIG 6A makes the 4-stroke a 2-stroke. To change to a 2-stroke from a 4-stroke, the cap 54 is partly unscrewed to a predetermined position on the engine 15, which raises shaft 43 and disengages idler 40A from sector gears 12 ~~(FIG 8A)~~ <u>(FIG 6A)</u>. The cap is held in place by known means, e.g. a dowel through the side of the cap that contacts engine 15.

~~Alternatively, for a 4-stroke, one or more dowels through engine 15 engage a radial groove in shaft 43 to prevent axial movement but allows rotary motion of idler 40A while engaging sector gears 12. To change to a 2-stroke, the dowels are removed from the groove. Idler 40A is separated~~

~~from sector gears 12 by lifting shaft 43 to where the dowels are inserted in a second groove. Shaft 43 is lifted to where the dowels are inserted in a second groove, which separates idler 40A from sector gear 12.~~

The second mechanical version is shown in FIG 12. Some reference numbers for the same parts in FIG 11 are omitted in FIG 12 to avoid overcrowding. ~~In FIG 12, the rod 101 is discarded by connecting one arm of the lever 100 directly to the wrist pin 97. A slant 25 of the contact surfaces is provided between the piston 81 and race 4. The spring 11 in FIG 13 can be included.~~ A lever 100 oscillates on its fulcrum 99 which extends from race 4. A gear mesh combines lever 100 with rod 84 to shift piston 81 into and out of contact with surface 112 on race 5. The piston is shown in contact with surface 112. The single piece rod 84 and piston 81 shift along a clutch radial 93 while in sliding contact with the carrying race 4. Space 88 allows the shift. Only a few teeth complete the gear mesh since the rod's motion is very short. A very short motion reduces backlash and may even make it negligible. If short enough, the gear mesh could be eliminated in favor of a single piece lever and rod. The spring-loaded trigger 85 at the end of arm 3 extends across gap 28 and stays in contact with the tough, long wearing strip 14 carried by race 5. The piston never contacts the strip 14. The trigger slides over strip 14 during overrun and grabs it at the beginning of the power stroke to oscillate the lever in response to the motion of race 5, thereby shifting the rod and piston. Torque is thus efficiently transmitted to race 4 perpendicular to the clutch radial 93.

Not shown is a third mechanical version that sets the piston on one radial of the clutch and the fulcrum on another. It can also eliminate the rod 101.

In all the 1-way clutch embodiments shown in FIGs 9-11,13: (1) the angle at the trigger's two extreme positions must not cause jamming, (2) the trigger should be suitably coated ~~with a suitable ceramic~~ and shaped to reduce drag but instantly grab the outer race when reversing to the drive direction, (3) the piston's motion 88 goes only far enough to provide clearance between the piston and the outer race during overrun and (4) one of the contact surfaces has a common V-groove and the other contact surface is beveled to fit it to prevent slip.

REMARKS

1. Sheets 2-4 show a marked up replacement paragraph consisting of math definitions and equations under my subsection heading, **Underlying Mathematics**.. Delete the several equations that have strikethrough lines, e.g ~~F = 16500(hp)(k)/π(n)(Rv)(r')~~ and insert all the underlined definitions and equations. The equations are followed by a newly added paragraph, which is followed by a marked up replacement paragraph consisting of an example that ends on sheet 4. The example is followed by strikethrough lines on sheets 4-6. Delete all the strikethrough lines, which will complete the entire subsection.

2. Sheets 7,8 of this 5th amendment begin with a newly added paragraph followed by two marked up replacement paragraphs. Insert the newly added paragraph followed by both replacement paragraphs into the specification immediately following the title, **Interchanging 4-Stroke and 2-Stroke.** and delete the entire 4th paragraph shown with strikethroughs through all its lines. That will replace my entire subsection.

3. Sheet 9 – The three paragraphs are the last three paragraphs in the written description. The two line unmarked paragraph between the two marked up replacement paragraphs is not new but is included for simplicity to avoid an error. Insert all three as shown as the last three paragraphs in the written specification.

4. The current status of claim 20. (currently amended): and claim 21 (currently amended): is used upon advice from the PTO in a phone inquiry. Claim 7. (thrice amended): is unchanged.

5. FIG 12 in the drawings has been replaced in the following sheet, which has "REPLACEMENT SHEET' printed in its top margin. None of the other drawings have been altered in this 5th amendment.

The matter in this 5th amendment is equivalent to existing matter. There is no new matter.

R.L. Giuliani
Inventor/Applicant

REPLACEMENT SHEET

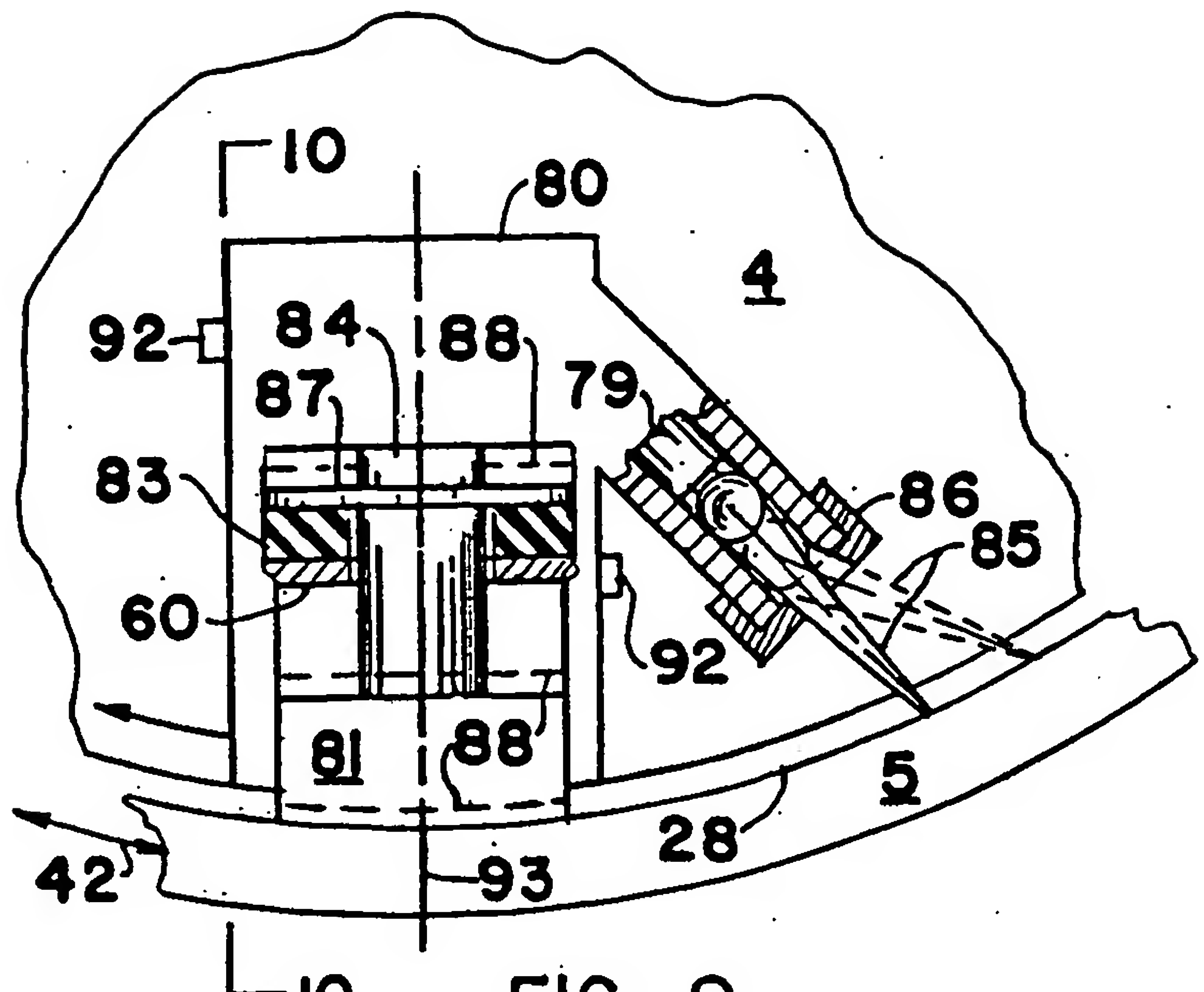

FIG. 9

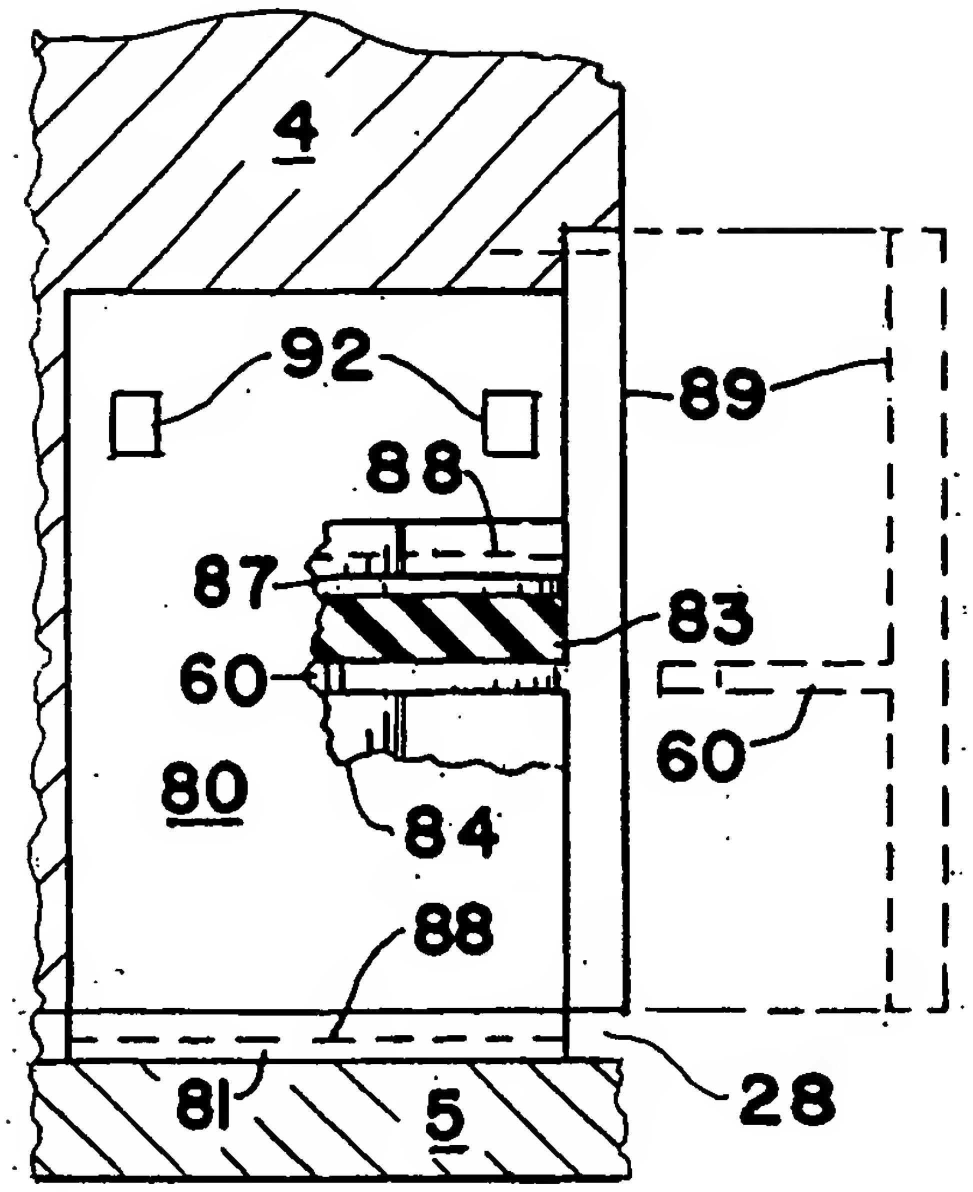

FIG. 10

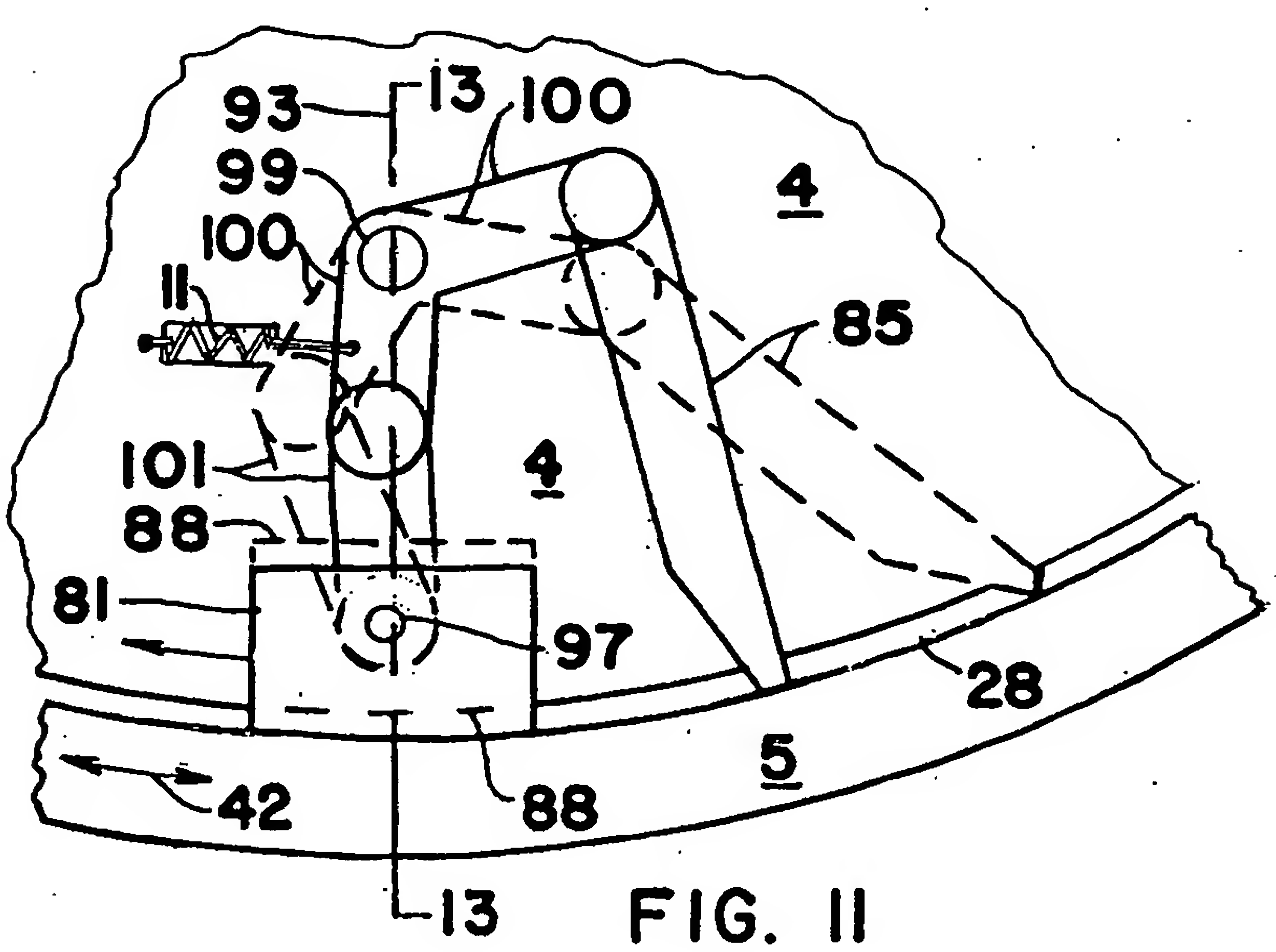

FIG. 11

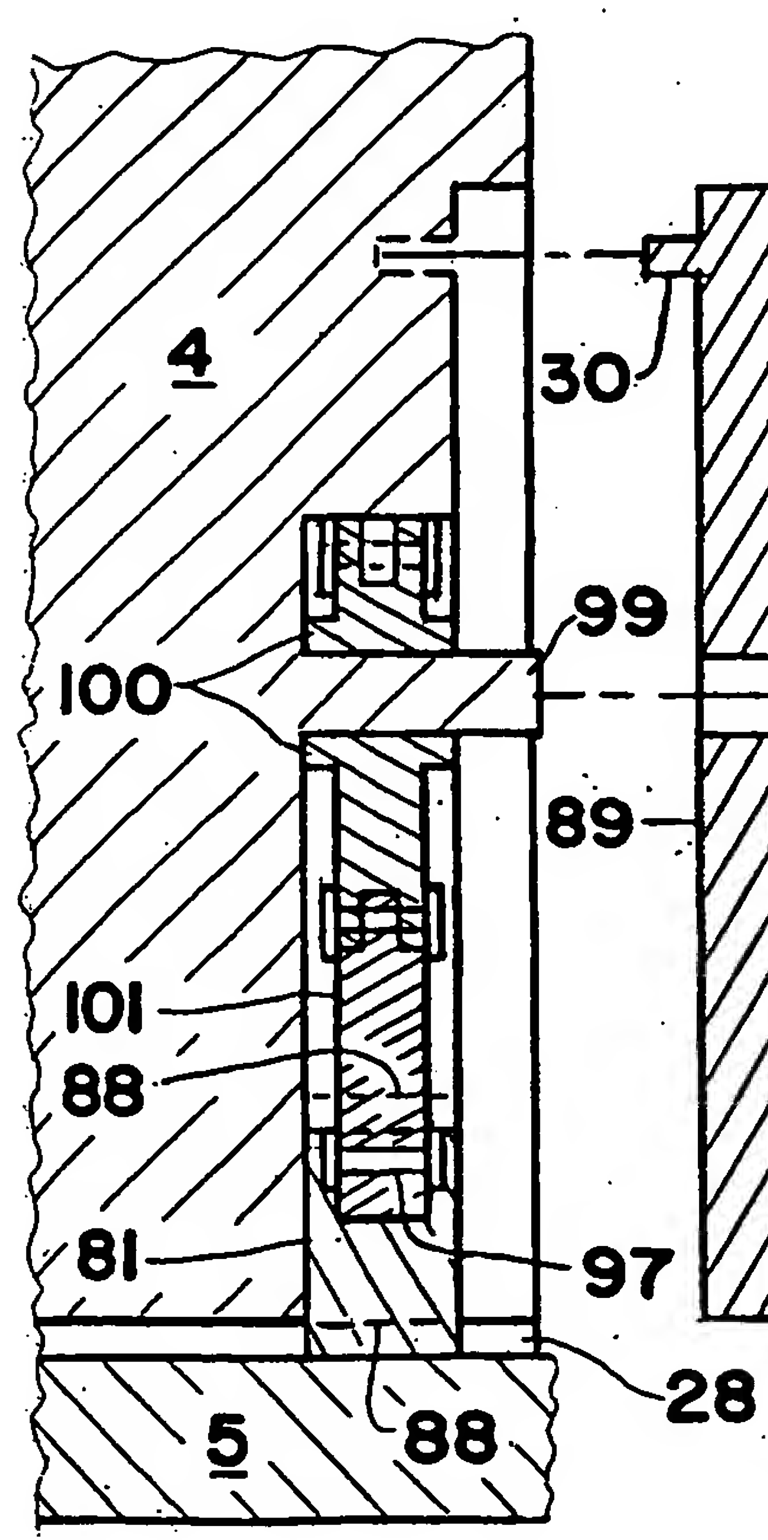

FIG. 13

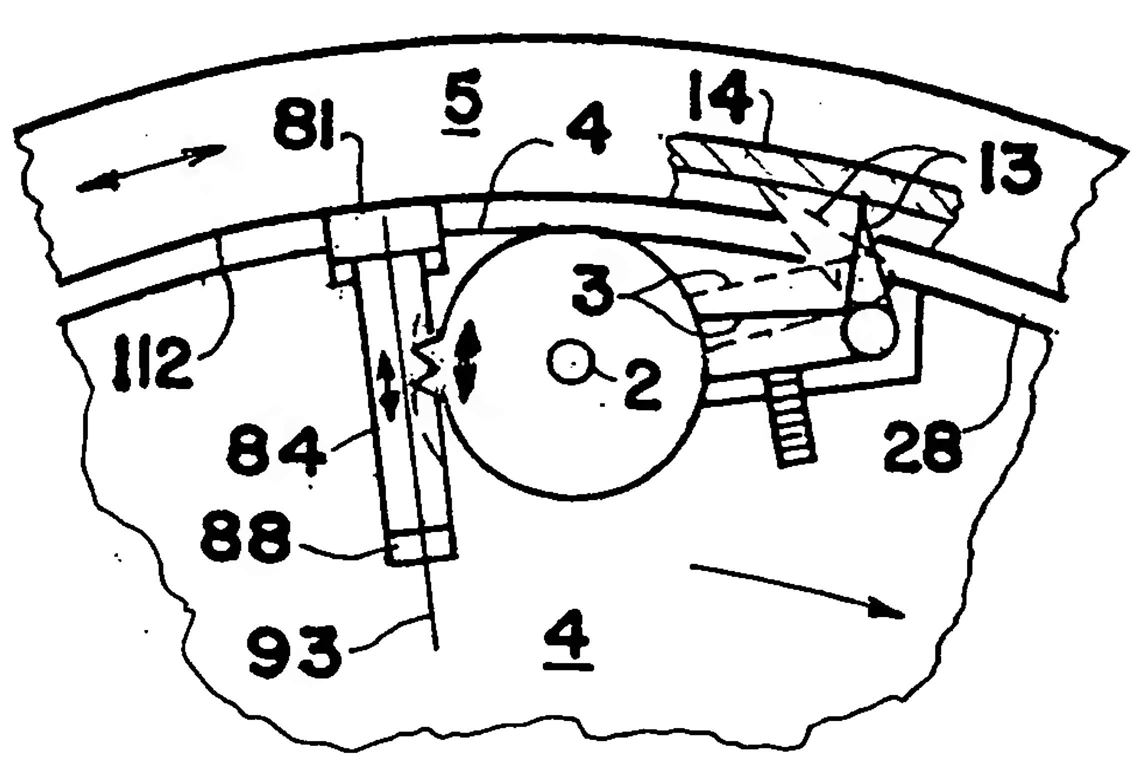

FIG. 12